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RESEARCH MEMORANDUM

EFFECT OF ANNULAR INLET BAFFLES ON ROTATING STALL, BLADE VIBRATION, AND PERFORMANCE OF AN AXIAL-FLOW COMPRESSOR IN A TURBOJET ENGINE

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SUMMARY

The effects of five different sizes of annular inlet tip baffles on rotating stall, blade vibration, and performance of a compressor in a J47-23 turbojet engine were investigated. The 5-percent-area tip baffle was sufficient to eliminate sustained rotating stall in the engine. Vibratory stresses in a second-stage rotor blade above the fatigue strength during operation with no baffle and with the exhaust nozzle closed to simulate rapid acceleration were reduced to safe levels with the installation of the 5-percent-area tip baffle. Compressor performance parameters were reduced by a factor of 1 to 4 percent by the 5-percent baffle at 65 percent of rated speed.

INTRODUCTION

Serious vibration problems due to rotating stall are encountered more and more frequently in axial-flow-compressor operation. When an axial-flow compressor is operated between 50 and 70 percent of rated speed, the ratio of inlet to outlet area in the compressor is inconsistent with the developed pressure ratio. A pattern of regions of high and low axial flow may be established in the annulus. These regions may be at the compressor case or at the hub. They may be large, each region covering several blades circumferentially and extending radially over the whole span of the blade, or small, covering only one blade for 5 to 10 percent of its span; and they may extend axially throughout the compressor. There may be any number of regions, and the whole pattern rotates in the direction of compressor rotation at roughly 50 percent of the rotor speed.

Rotating stall is a potential problem in two respects. It may impair the performance of the compressor, and it may cause serious resonant vibrations in the compressor blading. For example, in reference 1, rotating stall was directly responsible for exciting blade vibrations of sufficient magnitude to fail the stator blades by fatigue after a few

minutes of operation in an experimental compressor. Considerable work has been done on various turbojet engines with axial-flow compressors instrumented to measure strains in rotor or stator blades, or both (refs. 2 to 5). In most cases, blade vibrations excited by rotating stall were encountered; these vibrations were of sufficient magnitude to fail blades by fatigue.

There are several methods by which the size of the stalled regions and the magnitude of the velocity fluctuations might be reduced or the periodicity of the stall patterns might be broken up. One way of diminishing the effects of rotating stall is to vary the angle of the inlet guide vanes. The inlet-outlet area ratio could be made compatible with the pressure ratio by a partial blocking of the inlet. Since rotating stall is usually predominant at the base or tip of the blades, an annular hub or tip baffle might be introduced at the inlet of the compressor (refs. 5 and 6). Thus, the rotating stall and associated vibratory stresses may be reduced to a safe level.

The object of this investigation was to determine the effect of annular tip inlet baffles on performance, rotating stall, and associated blade vibrations in the compressor of a J47-23 turbojet engine. The engine was run under static sea-level conditions with no baffle and with five baffles of different sizes.

APPARATUS

The engine used in this investigation was a J47-23 turbojet mounted in a static sea-level test stand. A bellmouth and bullet nose at the front of the engine provided a smooth approach of air flow to the inlet. A variable-area exhaust nozzle was used. Temperatures and static and total pressures were measured with rakes and probes at the inlet and outlet of the compressor and at the tail cone. The temperatures were read on a self-balancing potentiometer, and the pressures were photographically recorded from banks of manometer tubes.

Two first-stage, four second-stage, and two third-stage compressor rotor blades were instrumented with commercial resistance-wire strain gages. The electrical circuit was completed to the rotor by means of slip rings, and power was supplied to the strain-gage bridges by storage batteries.

Hot-wire probes were supported by actuators whose radial positions were remotely controlled and indicated. The actuators were mounted in the compressor case between stator blades in the first and fourth stages. The output of the hot-wire probes was amplified by an NACA constant-current hot-wire amplifier (ref. 7). The hot-wire filament was of 0.001-inch-diameter wire 0.1 inch long. Radial filaments were used to

accommodate a wide variation in flow direction. The outputs from the strain gages and hot-wire anemometers were recorded on a 12-channel recording oscillograph and were monitored on a dual-beam cathode-ray oscilloscope. In addition, radial traverses with the hot-wire anemometers were recorded on a self-balancing potentiometer.

The two extreme baffle configurations are shown in figure 1. The baffle sizes investigated were 5, 10, 20, 30, and 40 percent of annulus area.

PROCEDURE

Performance data were taken at 50 percent of rated speed and at speed increments of 10 percent to rated speed. Because of temperature limitations, rated area could not be obtained at high speeds with baffles. Therefore, all performance data were taken with the nozzle open.

Hot-wire surveys in the first- and fourth-stage stator rows were made for each baffle configuration. The output from the hot-wire anemometer was passed through a circuit whose output was a d-c voltage proportional to the root mean square of the fluctuating input. This d-c output was then recorded on a self-balancing-potentiometer chart recorder. Once the desired engine conditions were set, the probe was moved radially inward at a constant speed by the actuator. At the same time, the chart recorder was turned on. Thus, a continuous radial traverse of the average air fluctuation was obtained from the compressor case to the rotor hub. Surveys were made at various speed increments from about 50 to 82 percent of rated speed (4000 to 6500 rpm) with the exhaust nozzle open and to 76 percent of rated speed (6030 rpm) with the nozzle set to give limiting exhaust temperatures at 76-percent rated speed.

Oscillograph records of rotating stall (bypassing the rms circuit) or of vibrations, or both, were taken whenever the monitoring oscilloscope showed them to be of particular interest. Care was taken to detect rotating stall in the speed range in which it is usually experienced in normal engine operation.

RESULTS AND DISCUSSION

Rotating Stall

A typical oscillogram of the output of a hot-wire anemometer when rotating stall is present in the compressor is shown in figure 2(a). The height of the peaks is a measure of the strength of the air fluctuation, and the abscissa is time. The peaks occurred at regular intervals. When the 5-percent baffle was installed, the hot-wire-anemometer output

appeared as in figure 2(b). The periodic nature of the peaks has disappeared. It is this periodicity of the stall pattern (fig. 2(a)) that is responsible for much compressor blade vibration.

Figures 3(a) and (b) are some typical radial traverses showing the variation in the radial direction of the rms magnitude of air fluctuation in the first-stage-stator annulus. Except where noted in the figures, the fluctuations shown were random; that is, rotating stall was not present. Figure 3(a) shows the fluctuation patterns with no baffle and with the exhaust nozzle open. At 4840 rpm, air fluctuation takes place over about one-fourth of the blade span. From 5240 to 5630 rpm, rotating stall was present in the compressor; the peaks of these curves are higher and narrower. However, this condition is not necessarily an indication of rotating stall. The stalled area at 5240 and 5360 rpm covered about 20 percent of the annulus. The upper limit of rotating stall was approached at 71 percent of rated speed (5630 rpm). Here, the averaged velocity fluctuation was less, as shown by the lower peak, and covered only about 5 percent of the blade span. Similar data were obtained with a rated nozzle (fig. 3(b)). The stall pattern tends to become weaker and to cover less of the blade as speed is increased. For this engine, then, it can be concluded that as the speed increases the peak strength of the stalls decreases and they extend over less of the annulus.

The effect of inlet baffles on magnitude of fluctuation is shown in figure 4. Increasing baffle size increases considerably the amount of the annulus covered by the fluctuation. As larger and larger baffles were installed, the peak moved toward the hub and decreased slightly in value (fig. 4(a)). Similar results are shown in figure 4(b), which presents data obtained for all baffles at 6000 rpm, at which speed there is normally no rotating stall. The peak amount of turbulence or air fluctuation was about doubled and the amount of annulus covered by the turbulence was increased by a factor of 4 with the introduction of the 5-percent baffle.

Figure 5 shows the effect of engine speed on the air fluctuation in the first-stage-stator annulus with the 5-percent baffle installed. The data shown in figure 5(a) were obtained with the exhaust nozzle closed to give limiting exhaust-gas temperature at 6030 rpm. The data of figure 5(b) were obtained with the nozzle open. These figures show that the peak air fluctuation moves toward the compressor case as the engine speed is increased. If the baffle is compared with an orifice, it might be expected that as the compressor speed is increased the vena contracta would move upstream toward the baffle and increase in diameter because of the increase in mass flow. The baffles had a similar effect on the air fluctuation in the fourth stage.

Vibratory Stress

Figure 6 shows vibratory stress in a second-stage blade, the most serious vibrations encountered in the engine without an inlet baffle. In reference 8 the complete map of the rotating-stall areas was determined, so that the stall patterns present during different acceleration paths could be predicted. At 5200 rpm with the exhaust nozzle closed beyond rated area to simulate a particular acceleration path, vibratory stresses above the fatigue strength were recorded. This stress was due to resonance between the second-stage blade frequency and the fundamental of the seven-zone stall frequency relative to the blades. Stresses encountered in the same blade with the 5-percent baffle installed are presented for comparison. The effect of the baffle can readily be seen. The vibratory stress was lowered from a stress above the fatigue strength to a safe operating stress.

Compressor Performance

The effects of the various baffle configurations on compressor performance with the exhaust nozzle open are shown in figure 7. In actual practice, the baffle would only be in position at about 5200 rpm or about 65 percent of rated speed. At this speed and with the nozzle open, the 5-percent baffle caused a drop of about 1 percent in efficiency, 2 percent in pressure ratio, and 4 percent in weight flow.

CONCLUSIONS

From an analysis of the data obtained in this investigation, the following conclusions may be drawn for the J47-23 engine considered in this report:

1. A 5-percent inlet baffle was sufficient to eliminate sustained rotating stall, although nonperiodic air fluctuations of larger magnitude were sometimes encountered.

2. The maximum vibratory stress encountered in the compressor with no inlet baffle installed was in the second stage, where a stress well above the fatigue strength was recorded when the exhaust nozzle was closed to simulate a particular acceleration path. This stress was reduced to a safe operating level with the installation of the 5-percent baffle.

3. A 5-percent inlet baffle reduced compressor efficiency 1 percent, pressure ratio 2 percent, and weight flow 4 percent at 65-percent rated speed with the exhaust nozzle open.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, March 28, 1955

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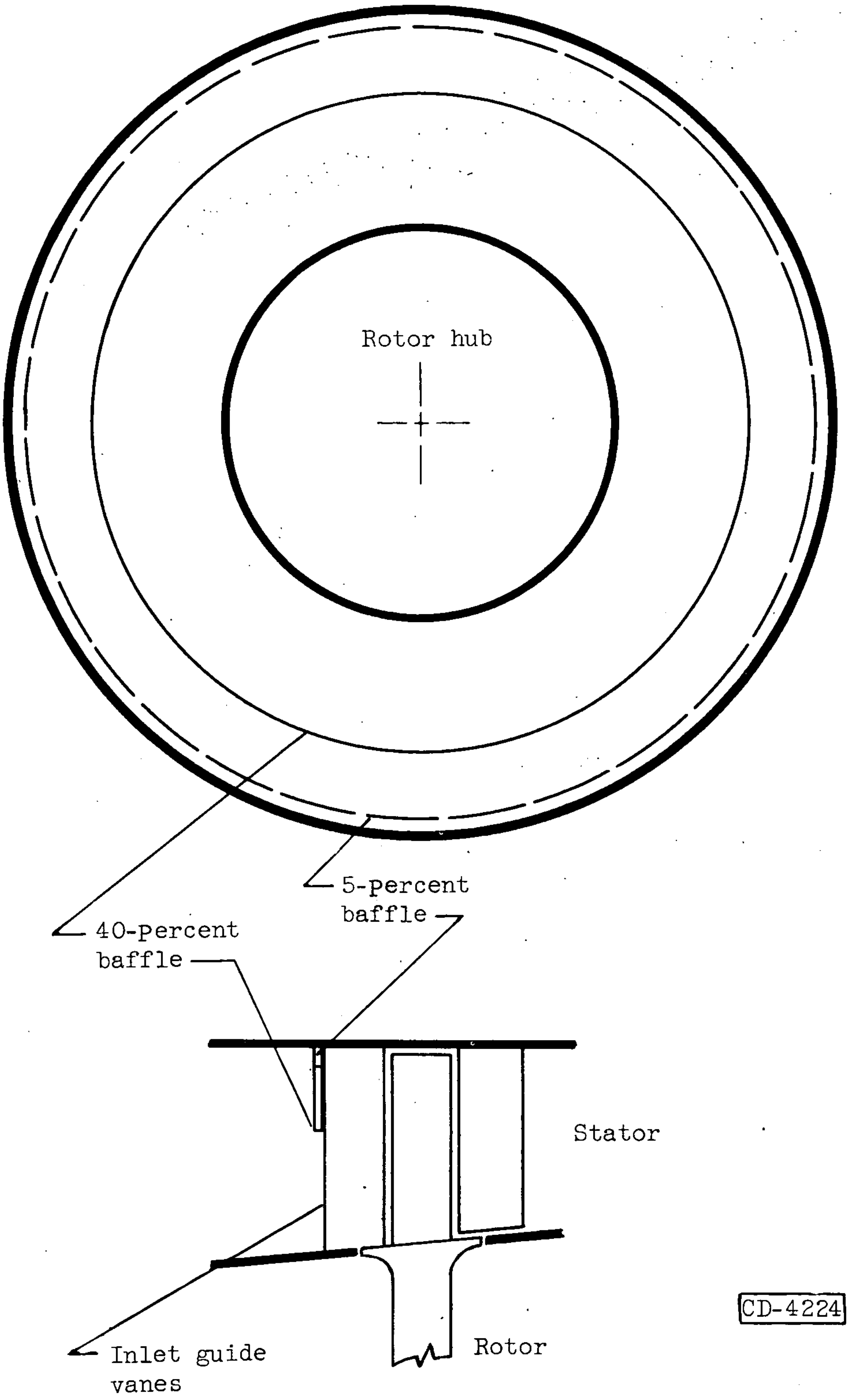
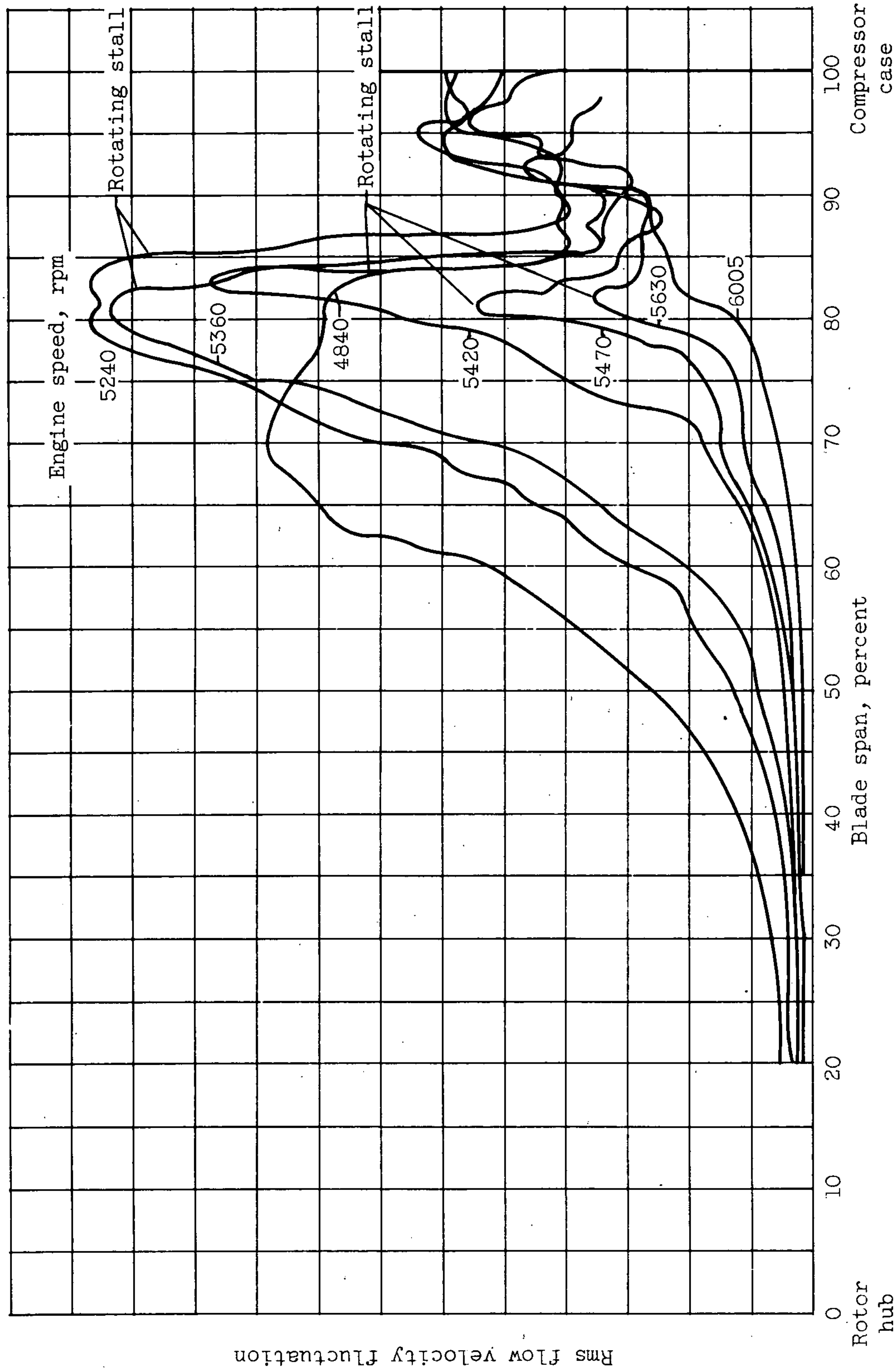


Figure 1. - Inlet-air baffles.

(a) Typical output of hot-wire anemometer when rotating stall is present.

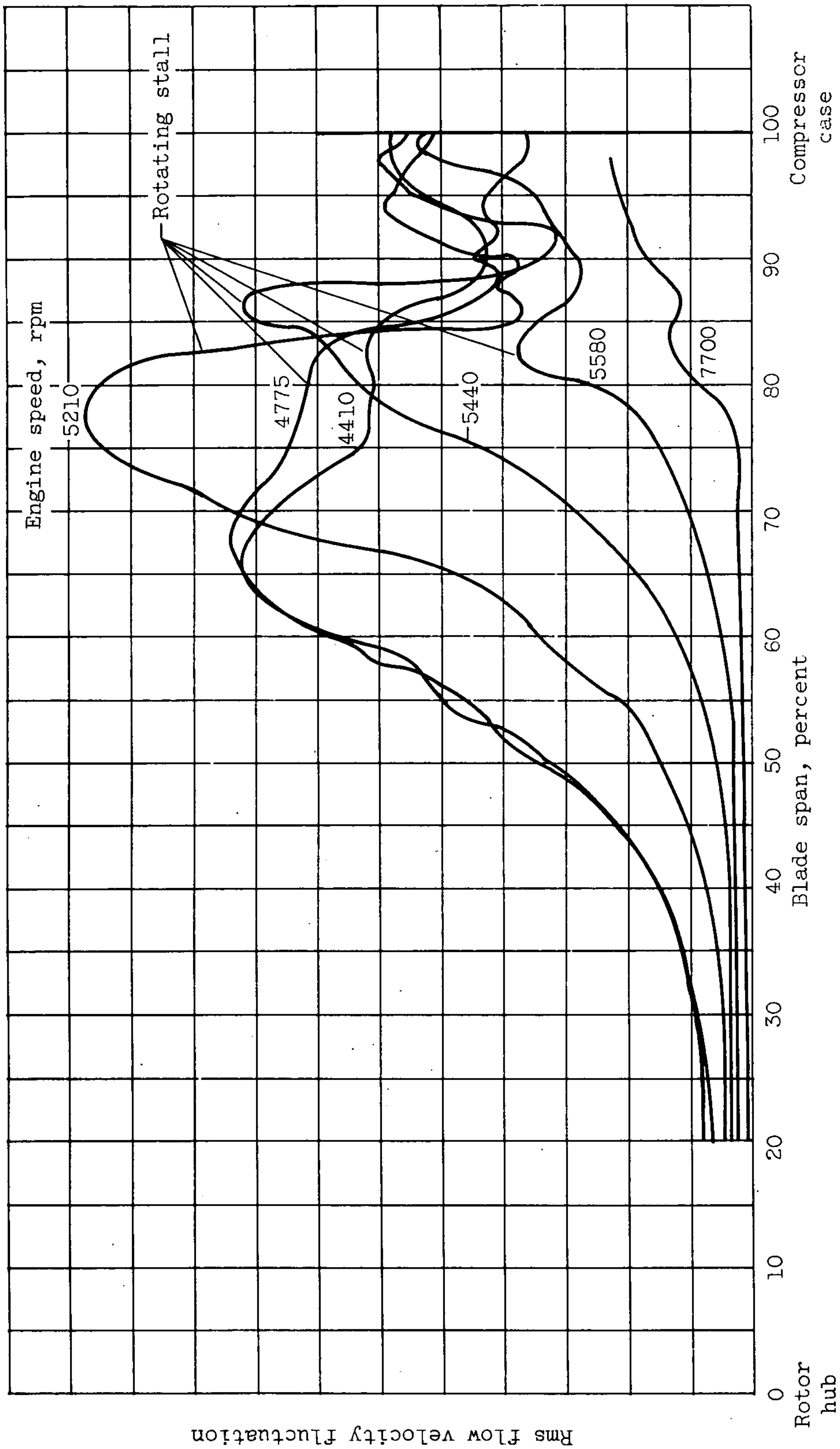
(b) Hot-wire-anemometer output with 5-percent baffle.

Figure 2. - Oscillograms of air fluctuations in first-stage stators at 5300 rpm.



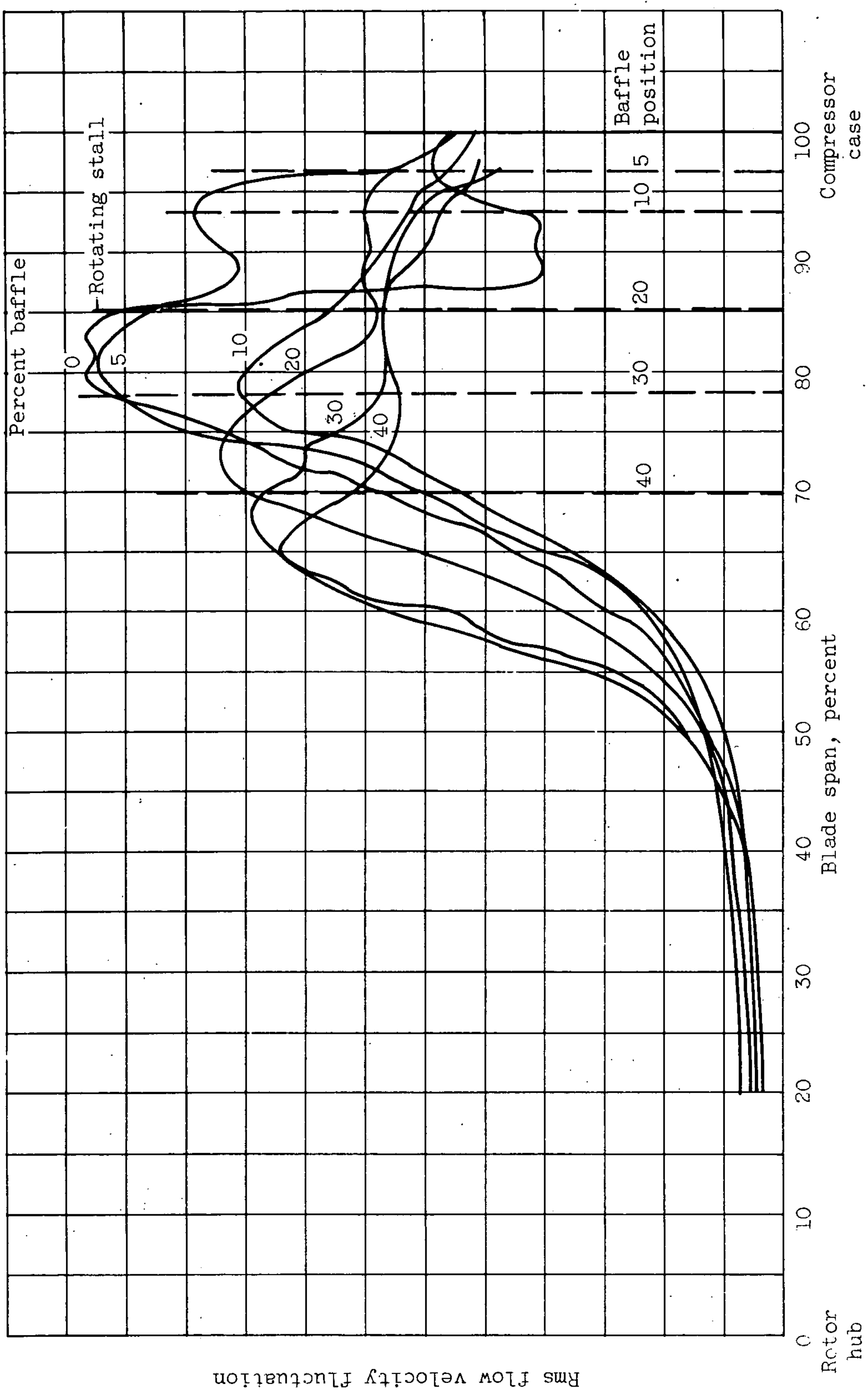
(a) No baffle; exhaust nozzle open.

Figure 3. - Radial traverses in first-stage-stator annulus.



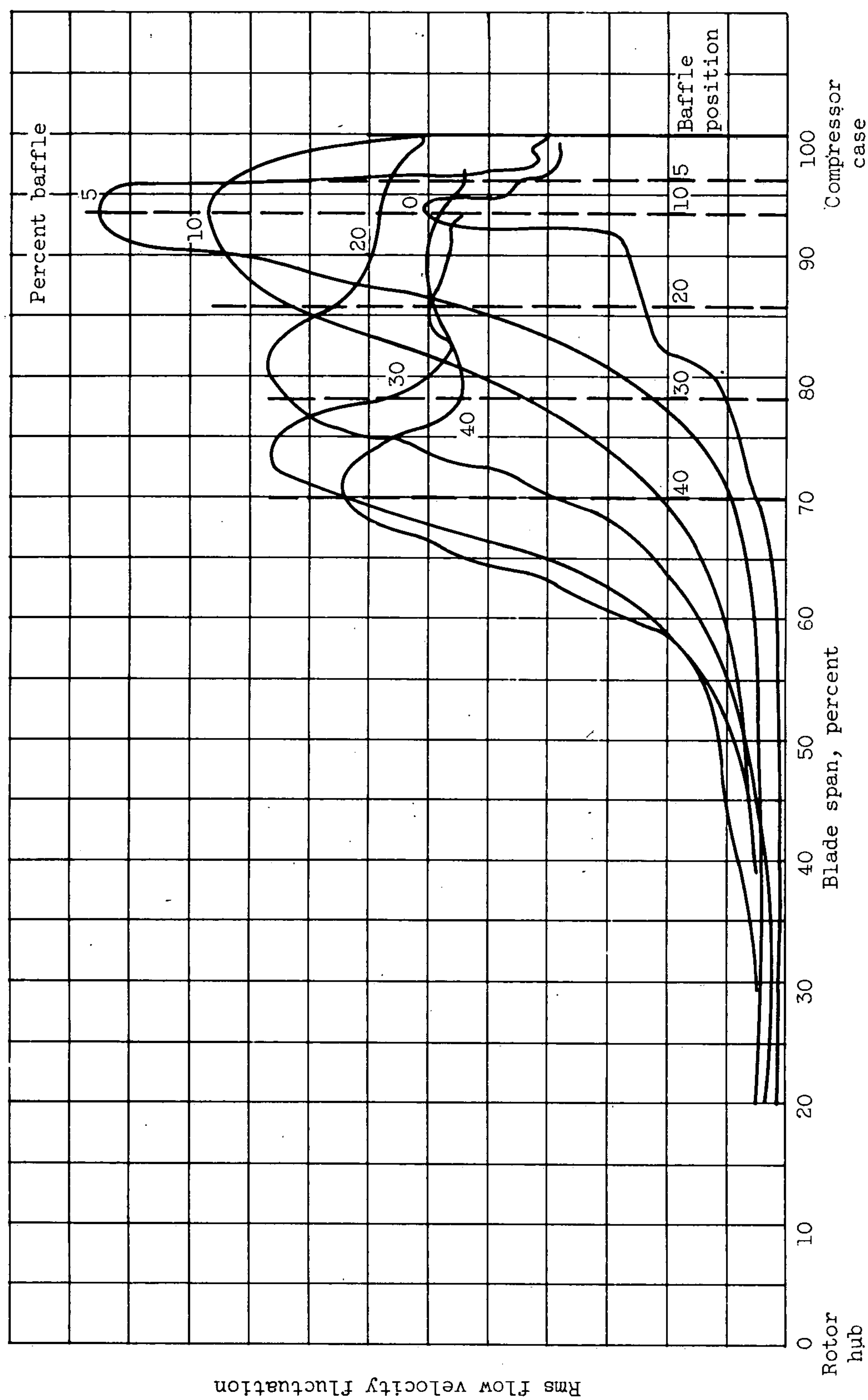
(b) No baffle; rated exhaust-nozzle area.

Figure 3. - Concluded. Radial traverses in first-stage-stator annulus.



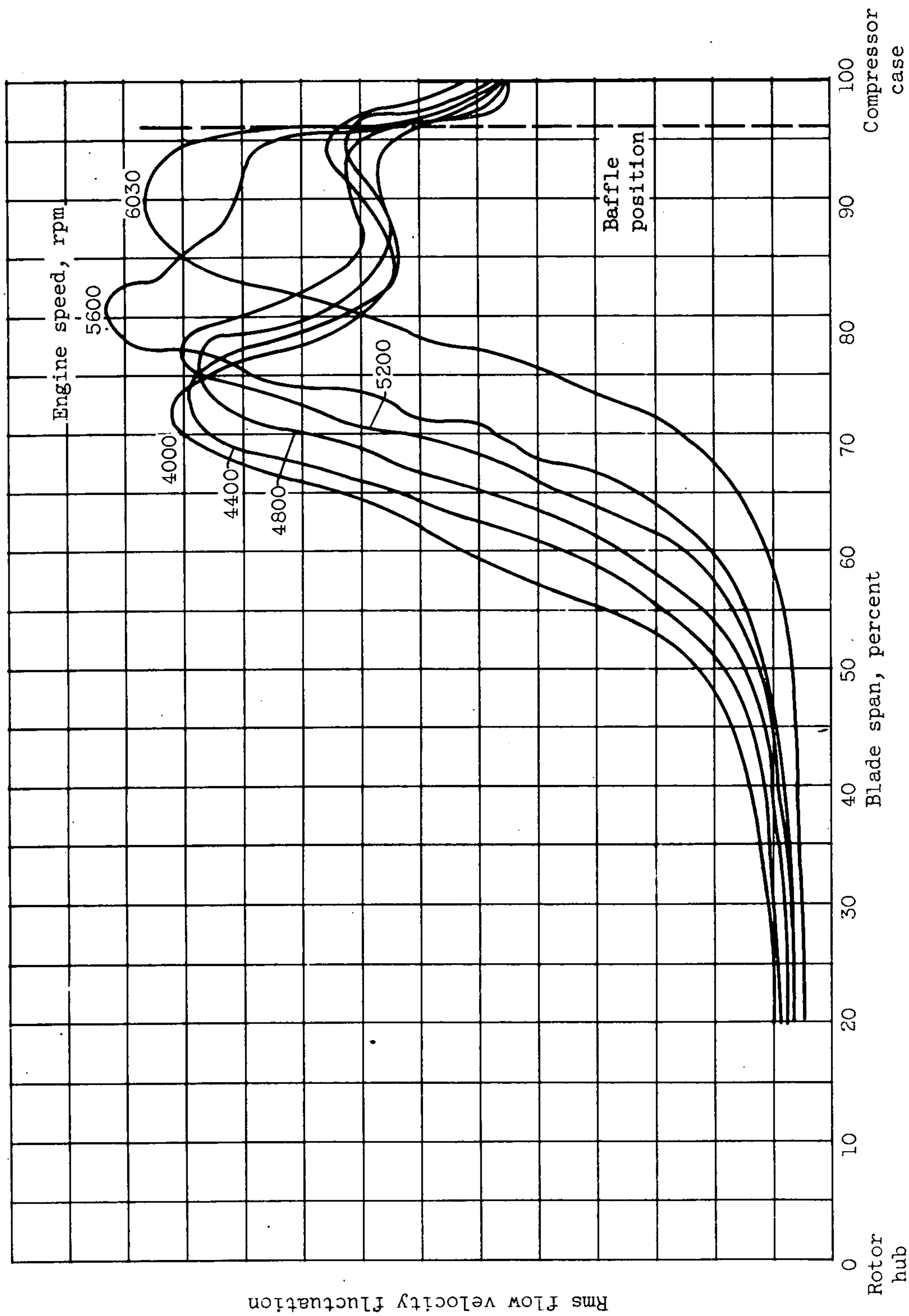
(a) Speed, 64.4-percent rated.

Figure 4. - Effect of baffle size on amount of annulus covered by fluctuation in first-stage-stator annulus.



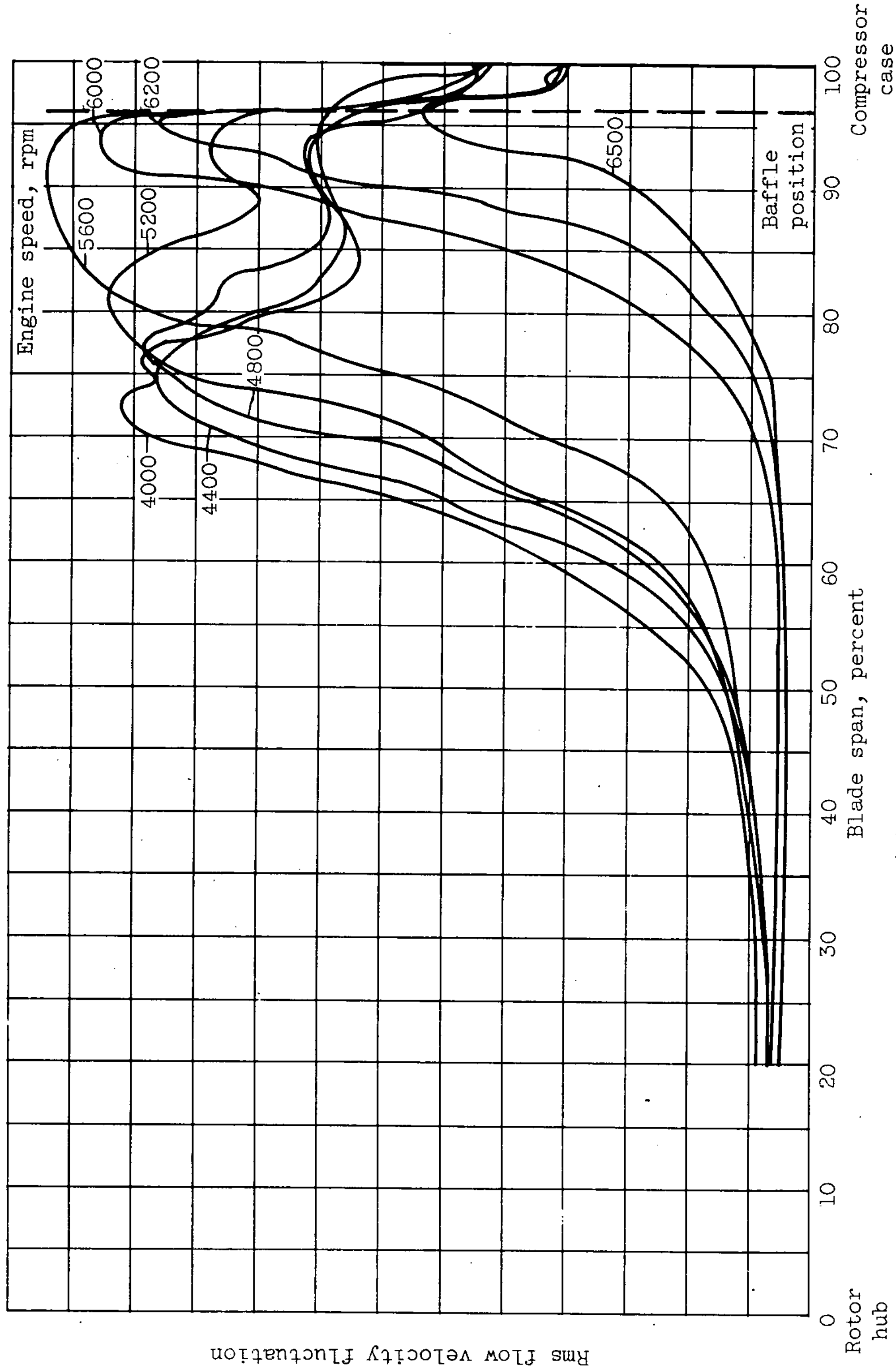
(b) Speed, 75.5-percent rated.

Figure 4. - Concluded. Effect of baffle size on amount of annulus covered by fluctuation in first-stage-stator annulus.



(a) Exhaust nozzle closed for exhaust-gas temperature of 1250° F at 76-percent rated speed.

Figure 5. - Effect of engine speed on air fluctuation in first-stage-stator annulus with 5-percent baffle.



(b) Exhaust nozzle open.

Figure 5. - Concluded. Effect of engine speed on air fluctuation in first-stage-stator annulus with 5-percent baffle.

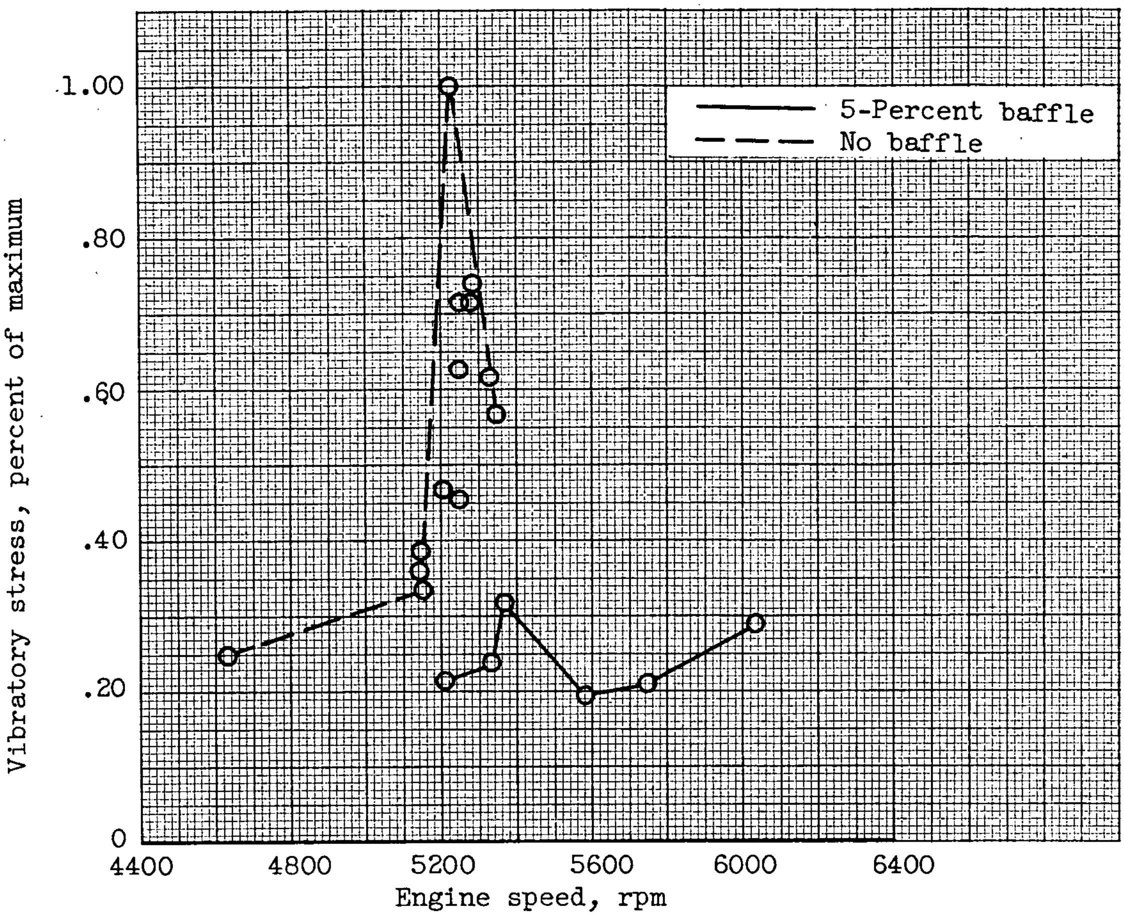
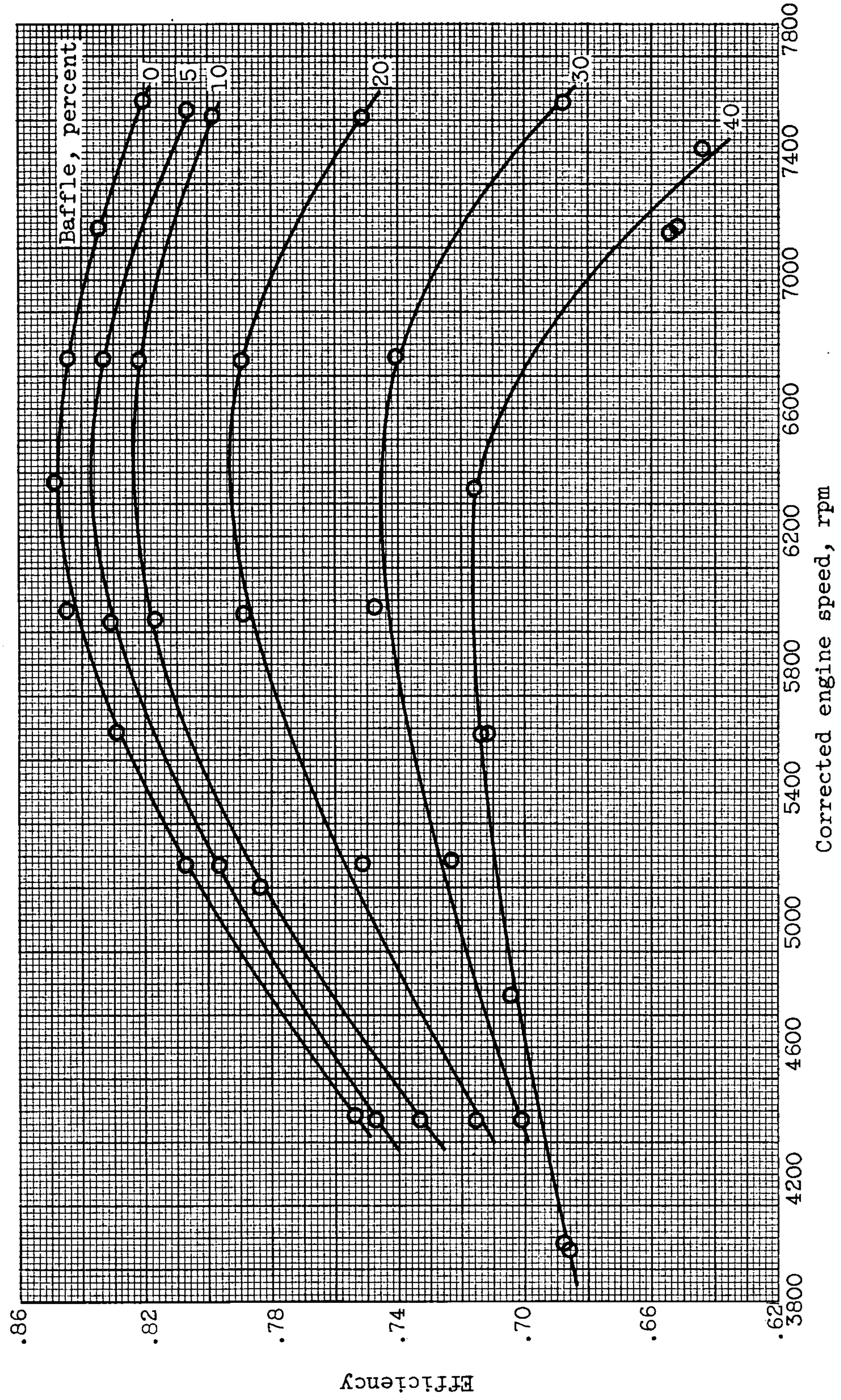
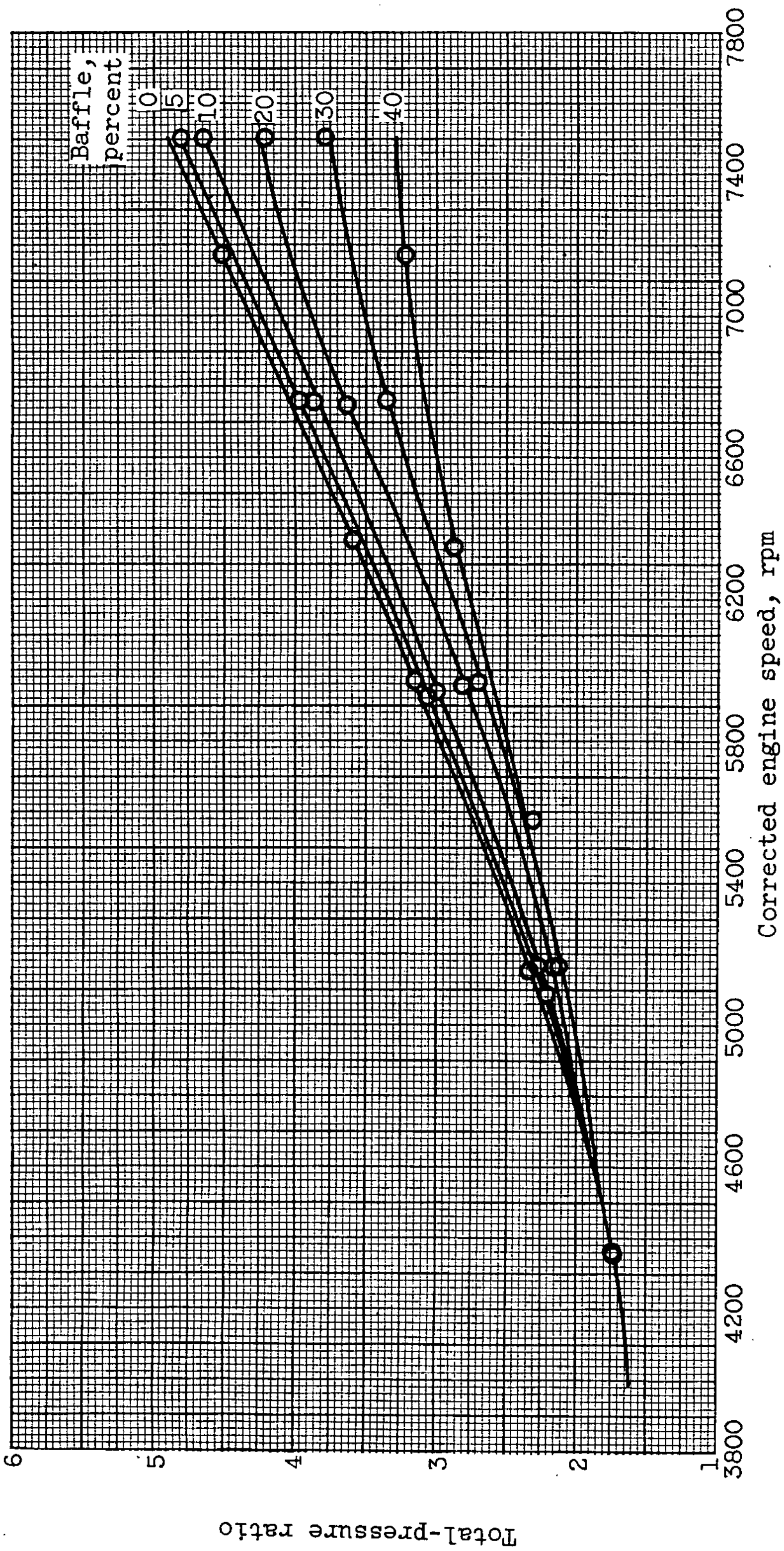


Figure 6. - Effect of baffle on second-stage vibratory stress. Exhaust nozzle closed for exhaust-gas temperature of 1250° F at 76-percent rated speed.



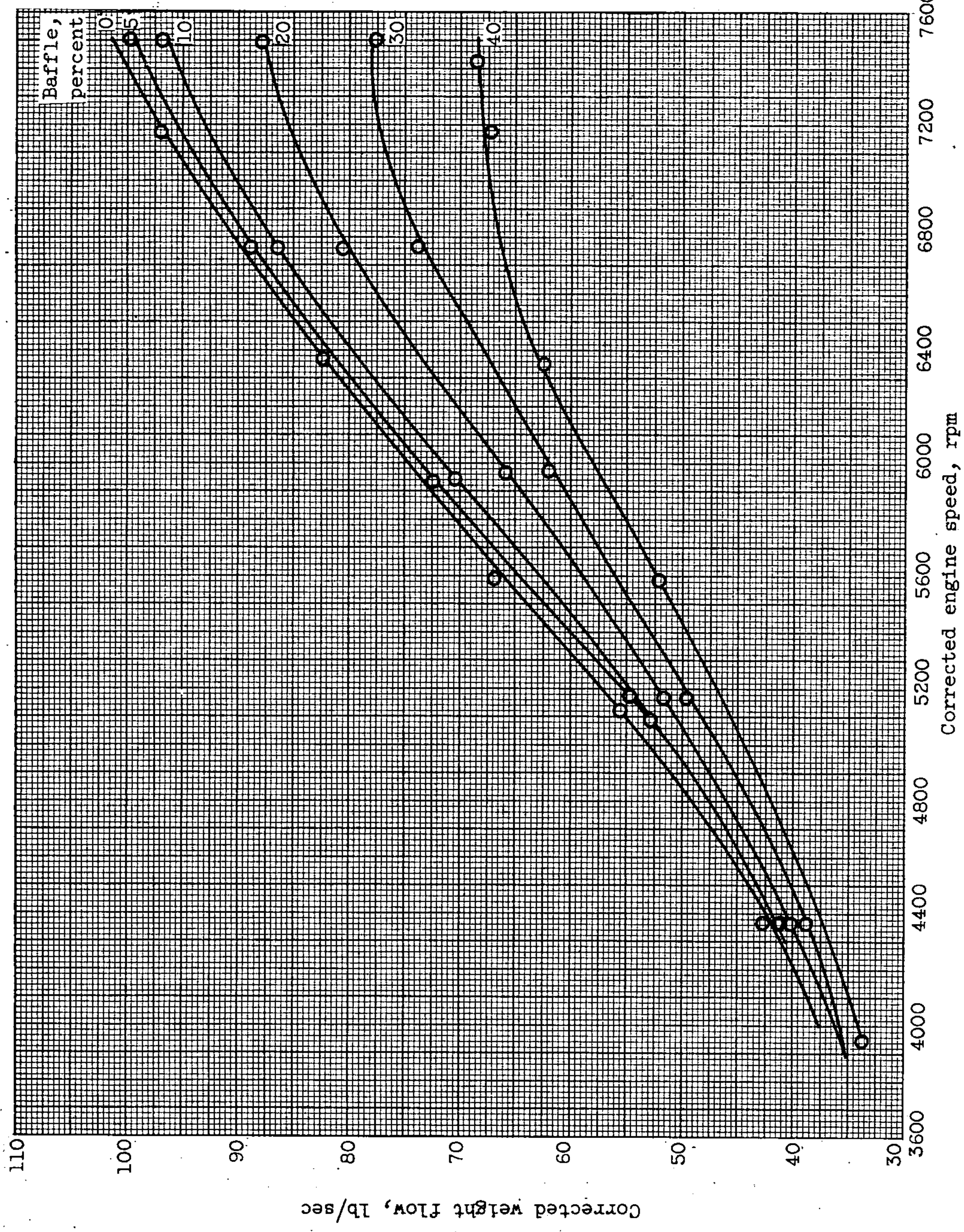
(a) Efficiency.

Figure 7. - Effect of various baffle configurations on compressor performance. Exhaust nozzle open.



(b) Pressure ratio.

Figure 7. - Continued. Effect of various baffle configurations on compressor performance. Exhaust nozzle open.



(c) Weight flow.

Figure 7. - Concluded. Effect of various baffle configurations on compressor performance. Exhaust nozzle open.

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